Effects of Commercial Kitchen Pressure on Exhaust System Performance

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ABSTRACT

This paper investigates the impact that the static pressure (i.e., positive, neutral, or negative pressure with respect to ambient) within a commercial kitchen space may have on exhaust system performance, since in real kitchens this pressure may change for a variety of reasons. Within a laboratory environment, it was demonstrated that conditions of positive pressure did not impact hood capture while conditions of negative pressure adversely affected the ability of the exhaust hood to capture and contain cooking effluent. This paper explains that this adverse effect was due to a reduction in the exhaust airflow rate caused by an increase in static pressure of the exhaust system, not the static pressure condition itself. The paper presents the theory that supports this conclusion as well as implications for facility design and operation.

BACKGROUND

Commercial kitchen ventilation systems are composed of many components. Exhaust hoods, exhaust fans, make-up air units, and packaged rooftop HVAC units all need to operate within specified tolerances to maintain optimum performance. The ability of exhaust hoods to capture and contain the effluent produced by cooking equipment is a fundamental criterion for the satisfactory performance of a commercial kitchen ventilation system. It is generally recognized that the factors affecting this performance include the style and design of the exhaust hoods, configuration and types of appliances under the exhaust hoods, the exhaust airflow rates, and the strategy for introducing replacement (makeup) air to the kitchen and/or exhaust hoods.

Another variable that is understood by designers and engineers, and to a lesser extent operators, as having an impact on the HVAC system performance is the pressure of the kitchen relative to other areas of the building and to the ambient (i.e., outside the building). Current design practice is for kitchens to be slightly negative with respect to the dining areas in order to contain cooking odors. Conventional design practice also calls for the pressure of the overall facility to be slightly positive with respect to ambient in order to minimize infiltration of dust, insects, and other outside contaminants.

In reality, the specified design pressure conditions may not be satisfied, as the actual airflow through kitchen hoods and replacement air systems may fall short of the specified quantities, particularly if a test and balance (T&B) report was not required as part of the facility start-up. And in real-world operation, the kitchen may experience further pressure imbalances as intake filters and blower wheels load with dirt, ductwork becomes disconnected, fan belts slip or break, and economizer dampers fail in an open or closed position.

The most familiar example is the experience of feeling resistance to opening outside doors, which suggests that the overall facility is negative with respect to ambient. On the other end of the spectrum, the smell of kitchen odors may permeate dining areas, indicating that the kitchen is positive with respect to dining areas or there are problems with hood capture and containment. Economizers with passive pressure relief may cause positive pressure conditions that are not easily perceived. Severe positive pressure conditions in the kitchen are less likely to occur, as this would typically be caused by an exhaust fan failure that is more quickly perceived by kitchen staff.

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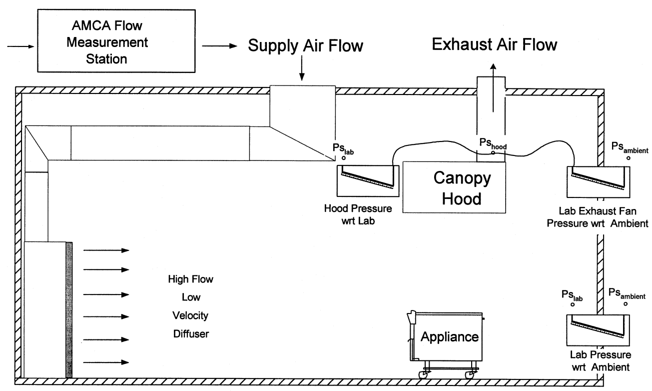

Figure 1 *Experimental setup.*

INTRODUCTION

Further to the background discussion, it is generally acknowledged and anecdotally reported that kitchen or building pressures can affect the capture and containment (C&C) performance of an exhaust hood. Although the authors of this paper have been extensively involved with performance testing of hoods under both field and laboratory conditions for many years, they were not able to identify studies that actually documented the effect of kitchen pressure on hood performance. Even more specific to the authors' experience was the fact that they themselves had not investigated this effect within the scope of ongoing research at a commercial kitchen ventilation testing facility.

This paper investigates negative, neutral, and positive pressures within a laboratory space simulating a commercial kitchen and the effects of these pressures on the capture and containment (C&C) performance of an exhaust hood, including the physical principles involved.

Hood Capture and Containment (C&C) Testing

There are laboratory methods of test to determine the threshold exhaust airflow rate for complete capture and containment for a particular hood/appliance system as described in ASTM 1704, *Standard Test Method for the Performance of Commercial Kitchen Ventilation Systems* (ASTM 1999). The testing is most conveniently performed in a relatively airtight laboratory, as described in Soling and Knapp (1985), Gordon et al. (1995), and Swierczyna et al. (1997). Within an airtight lab, the mass flow of supply and exhaust air remains constant. Other than the addition of mass (typically negligible) from the combustion of natural gas, the supply and exhaust mass flow rates are equal. Thus, the grease-laden and heated exhaust airstream need not be measured, as accurate measurement of the clean supply airflow will provide a determination of the exhaust flow rate. If supply and exhaust flows are varied with respect to each other, mass flow rate will still be equal, but a positive or negative pressure can be set with respect to ambient. This parameter can be conveniently and precisely measured. Additional pressure taps can also be provided to relate pressure differences at various points in the overall experimental system.

When a hood/appliance system is tested, the exhaust flow rate is determined by operating the equipment under typical operating conditions and varying the exhaust to confirm the threshold flow rate for complete capture and containment. The latter is determined by a flow visualization technique, such as a schlieren effect (Schmid et al. 1997). The speed of the variable-speed exhaust fan is then locked. The test appliance is cooled down and a neutral pressure is maintained between the lab and ambient. At that point, with temperatures equal, the supply flow is equal to the exhaust flow and an accurate measurement of the supply is taken as the "design" exhaust flow rate (cfm). In general practice, supply and exhaust flow rates are verified by test and balance upon commissioning of new systems, though test conditions may not match those in the laboratory. Variations may include changes of cooking equipment, temperature settings, and changes in food products, all of which may affect performance, but additionally, pressure conditions may not match those presumed or specified by designers. In general though, once exhaust flow rates are verified in facilities, the exhaust fan speeds are set and recorded.

Test Setup, Procedure, and Results

While investigating the threshold for capture and containment of two 3-ft (0.9 m) gas griddles idling under an 8-ft by 4-ft (2.4 m $\times$ 1.2 m) wall canopy hood, sensitivity testing (and the basis for this paper) led to varying the lab pressure with respect to ambient and determining the effect on capture and containment and flow rate. In order to do this, the lab was set up and instrumented as shown in Figure 1. The tests were run with similar supply and exhaust airflow densities; the volumetric flow entering was within 2% of the flow exhausted (although the mass flow was equal). Static pressures were measured and recorded at three points. These pressures included the lab static pressure, the hood static pressure (measured after the exhaust collar and near the fan inlet), and the ambient (outside) pressure. The lab static pressure, which is the variable of principal interest, was measured relative to the ambient condition. The hood static pressure was measured relative to the ambient condition and relative to the lab. The first set of conditions (Experiment I) included maintaining a constant speed on the exhaust fan and varying the supply fan speed to change the pressure in the lab. The results are shown in Table 1. The second set of conditions (Experiment II) included maintaining a constant supply flow rate and varying the exhaust fan speed to change the pressure in the lab. The results are shown in Table 2. In each case, the supply flow rate was taken to equal the exhaust flow rate.

TABLE 1a
Experiment I: Hood Performance with a Constant Exhaust Fan Speed and Varied Supply Fan Speed

Lab Pressure (i.e., lab wrt ambient), in. of water	Hood Pressure (i.e., hood wrt lab), in. of water	Lab Exhaust Fan Pressure (i.e., fan wrt ambient), in. of water	Supply Flow Rate, cfm	Supply Fan Speed, rpm	Exhaust Flow Rate, cfm	Hood Performance Capture & Containment
0.200	–0.186	–0.005	1990	2468	1990	C&C
0.100	–0.145	–0.055	1750	2170	1750	C&C
0.050	–0.128	–0.075	1585	1965	1585	C&C
0.000	–0.100	–0.100	1442	1788	1442	C&C
–0.010	–0.095	–0.105	1415	1755	1415	Spill
–0.020	–0.093	–0.110	1390	1724	1390	Spill
–0.030	–0.085	–0.115	1355	1680	1355	Spill
–0.050	–0.080	–0.125	1275	1581	1275	Spill
–0.100	–0.055	–0.155	1125	1395	1125	Spill
–0.180	–0.003	–0.170	560	694	560	Spill

TABLE 1b
Experiment I: Hood Performance with a Constant Exhaust Fan Speed and Varied Supply Fan Speed (SI Units)

Lab Pressure (i.e., lab wrt ambient), Pa	Hood Pressure (i.e., hood wrt lab), Pa	Lab Exhaust Fan Pressure (i.e., fan wrt ambient), Pa	Supply Flow Rate, L/s	Supply Fan Speed, rpm	Exhaust Flow Rate, L/s	Hood Performance Capture & Containment
49.8	–46.3	–1.2	939	2468	939	C&C
24.9	–36.1	–13.7	826	2170	826	C&C
12.5	–31.9	–18.7	748	1965	748	C&C
0.0	–24.9	–24.9	680	1788	680	C&C
–2.5	–23.7	–26.1	668	1755	668	Spill
–5.0	–23.2	–27.4	656	1724	656	Spill
–7.5	–21.2	–28.6	639	1680	639	Spill
–12.5	–19.9	–31.1	602	1581	602	Spill
–24.9	–13.7	–38.6	531	1395	531	Spill
–44.8	–0.7	–42.3	264	694	264	Spill

TABLE 2a
Experiment II: Hood Performance with a Constant Supply Flow Rate and Varied Exhaust Fan Speed

Lab Pressure (i.e., lab wrt ambient), in. of water	Hood Pressure (i.e., hood wrt lab), in. of water	Lab Exhaust Fan Pressure (i.e., fan wrt ambient), in. of water	Supply Flow Rate, cfm	Exhaust Flow Rate, cfm	Exhaust Fan Speed, rpm	Hood Performance Capture & Containment
0.200	–0.086	0.102	1450	1450	95	C&C
0.100	–0.095	–0.001	1450	1450	205	C&C
0.050	–0.095	–0.051	1450	1450	245	C&C
0.000	–0.102	–0.099	1450	1450	275	C&C
–0.010	–0.108	–0.111	1450	1450	285	C&C
–0.020	–0.107	–0.122	1450	1450	290	C&C
–0.030	–0.108	–0.134	1450	1450	300	C&C
–0.050	–0.115	–0.151	1450	1450	310	C&C
–0.100	–0.113	–0.203	1450	1450	340	C&C
–0.180	–0.117	–0.286	1450	1450	385	C&C
–0.200	–0.117	–0.303	1450	1450	395	C&C

TABLE 2b
Experiment II: Hood Performance with a Constant Supply Flow Rate and Varied Exhaust Fan Speed (SI Units)

Lab Pressure (i.e., lab wrt ambient), Pa	Hood Pressure (i.e., hood wrt lab), Pa	Lab Exhaust Fan Pressure (i.e., fan wrt ambient), Pa	Supply Flow Rate, L/s	Exhaust Flow Rate, L/s	Exhaust Fan Speed, rpm	Hood Performance Capture & Containment
49.8	–21.4	25.4	684	684	95	C&C
24.9	–23.7	–0.2	684	684	205	C&C
12.5	–23.7	–12.7	684	684	245	C&C
0.0	–25.4	–24.7	684	684	275	C&C
–2.5	–26.9	–27.6	684	684	285	C&C
–5.0	–26.6	–30.4	684	684	290	C&C
–7.5	–26.9	–33.4	684	684	300	C&C
–12.5	–28.6	–37.6	684	684	310	C&C
–24.9	–28.1	–50.5	684	684	340	C&C
–44.8	–29.1	–71.2	684	684	385	C&C
–49.8	–29.1	–75.4	684	684	395	C&C

For Experiment I, the exhaust fan speed was held constant (262 rpm) and the supply fan speed varied to create different pressure (and resulting airflow) conditions (see Table 1). At a lab pressure of 0.000 in. of water, the supply fan speed was 1788 rpm and the flow through the lab was 1442 cfm (680 L/s). This was also the predetermined threshold airflow for complete C&C. To create a condition of +0.200 in. of water (49.8 Pa) pressure in the lab with the exhaust speed held constant, the supply fan speed was increased from 1788 rpm to 2468 rpm. The flow through the system increased from 1442 cfm (680 L/s) to 1990 cfm (939 L/s) as measured on the supply side.

To create a lab pressure of –0.180 in. of water (–44.8 Pa), the supply fan speed was decreased to 694 rpm, delivering only 560 cfm (264 L/s) through the system. This change in speed of the supply fan created a different pressure within the system as seen by the exhaust fan. This change in pressure induced by the supply fan created the varying system pressures (the lab pressure being one component) and the result was changing airflows through the supply/lab/exhaust system.

The hood demonstrated complete capture and containment under all cases where the pressure in the lab was positive and the airflow had increased above the threshold exhaust flow rate for C&C. The capture and containment performance

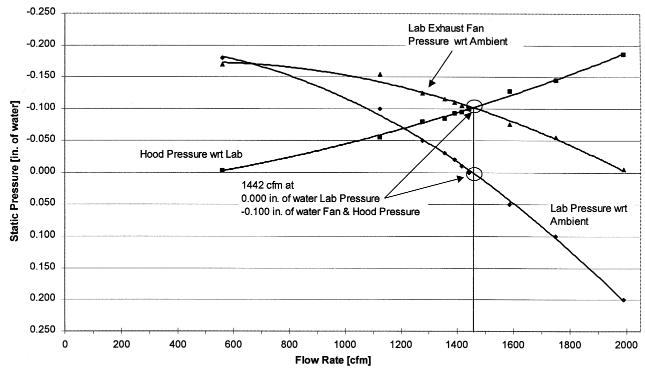

Figure 2a *Measured pressures and flow rates (Experiment I).*

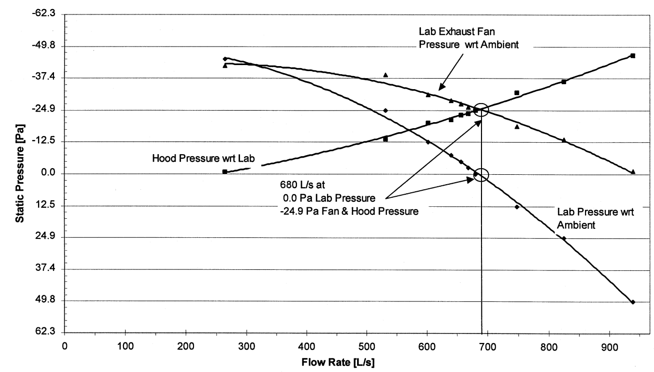

Figure 2b *Measured pressures and flow rates (Experiment I) (SI Units).*

of the hood failed when the pressure in the lab became negative, even as slight as 0.010 in. of water (2.5 Pa) with respect to the ambient, as the exhaust flow rate had fallen below the threshold for C&C.

To verify that the failure of the hood to capture and contain was only a function of the reduced exhaust rate caused by the negative pressure and not compounded by the negative pressure condition, Experiment II was conducted. In this case, the supply flow rate was held constant at 1450 cfm (684 L/s) and maintained through the system (Table 2). As in Experiment I, the airflow that entered the lab was the same as the airflow exhausted, established by the airtight condition. However, in this case, to vary the pressure in the lab the exhaust flow fan speed was varied. The pressure in the lab was changed from +0.200 in. of water (+49.8 Pa) to –0.200 in. of water (–49.8 Pa) by changing the exhaust fan speed from 395 rpm to 95 rpm, respectively. Regardless of the pressure condition, the flow through the exhaust remained constant and the capture and containment performance of the hood was good through the range of lab pressures tested.

Figure 2 illustrates the relationship among the measured system flows and ambient, lab, hood, and fan pressures for the various lab pressure scenarios at a constant exhaust fan speed (Experiment I). Changing the supply fan speed varied the pressure in the lab and, consequently, the flow through the supply/lab/exhaust system. As the lab pressure changed, the total exhaust system static pressure changed. The lab's exhaust system static pressure with respect to ambient is composed of the exhaust ductwork, elbows, hood, filters, and lab pressure induced by the supply fan. In a typical system design stage, a change in the system static curve is related to the change in the length of run, duct diameter, or fitting losses. In this study, a change in system static pressure was related to the change in lab pressure. Each of the points shown on the fan pressure curve represents a single point on the exhaust system static pressure curve, for a given lab pressure condition. The exhaust fan pressure curve is shown negative because it was measured on the inlet side of the fan.

Figure 2 also shows that the hood static pressure increases (i.e., the pressure drop across the hood increases in magnitude or becomes more negative) as the flow increases. However, the exhaust fan "sees" more than the pressure drop across the hood; it also operates against the lab pressure. The sum of the lab and hood pressures is the total pressure "seen" by the fan. The correlation is shown as "Lab Exhaust Fan Pressure" in Table 1 or Figure 2.

The 0.000 pressure condition on the lab pressure curve represents the pressure condition where the hood is typically tested in the lab for capture and containment. For the specific hood tested, 1442 cfm (680 L/s) was the established threshold capture and containment flow rate. The hood pressure at the 1442 cfm (680 L/s) flow rate was approximately 0.100 in. of water (24.9 Pa). The fan pressure is the sum of the lab and hood pressure conditions. The fan pressure (at the C&C tested condition) is then 0.000 in. of water lab pressure plus –0.100 in. of water (–24.9 Pa) hood pressure, which totals to –0.100 in. of water (–24.9 Pa) system or fan pressure. At the +0.200 in. of water (+49.8 Pa) lab pressure, the pressure drop across the hood was –0.186 in. of water (–46.3 Pa), which totaled to the fan pressure of –0.005 in. of water (–1.2 Pa). The fan pressure could then be related to the flow through the system. For example, the –0.005 in. of water (–1.2 Pa) fan pressure relates to the 1990 cfm (939 L/s) flow through the system. In the same way, at the –0.180 in. of water (–44.8 Pa) lab pressure, the pressure at the hood was –0.003 (–0.7 Pa) and the pressure measured at the exhaust fan was –0.170 in. of water (–42.3 Pa) and the flow through the system was 560 cfm (264 L/s). This pressure-flow correlation held true for each lab pressure condition tested.

The overall effect is that the negative lab pressures were added to the exhaust system's static pressure and decreased the flow through the system. However, when the lab pressure increased in the positive direction, this subtracted from the system's static pressure and increased the flow rate through the system. The key to understanding the overall effect is that the pressure at the fan as a function of flow rate is a typical fan pressure-flow curve for a given fan at a given speed. The

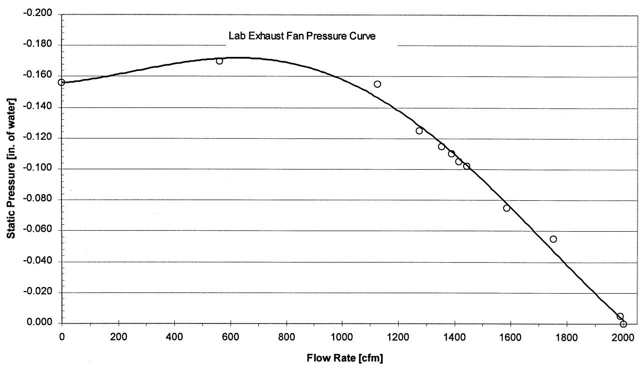

Figure 3a *Lab exhaust fan pressure curve (262 rpm).*

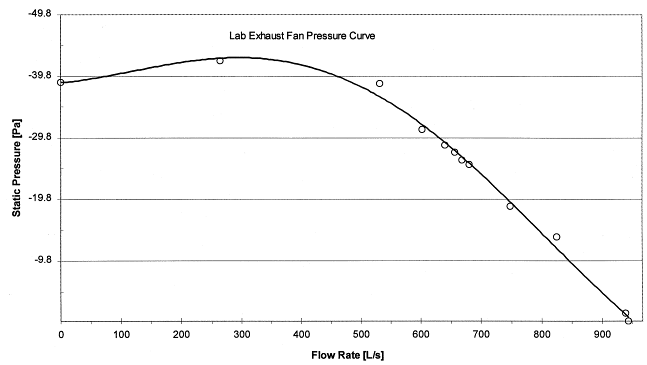

Figure 3b *Lab exhaust fan pressure curve (262 rpm) (SI Units).*

process was then repeated for nine other pressures and flow rates to establish Table 1 and the set of curves shown in Figure 2.

Figure 3 is a detail of the fan pressure-flow curve for the lab's exhaust fan. The negative pressures represent the pressures on the inlet side of the fan. Each point on the lab exhaust fan pressure curve was measured during the various lab pressures generated by the supply fan. Each point on the exhaust curve represents a change in the exhaust system's static pressure and, consequently, a change in flow on the fan curve at a constant fan speed of 262 rpm.

ANALYSIS

Figure 4 shows the effect of changing lab pressure on flow and consequent capture and containment. The operating point of 1442 cfm (680 L/s) on the fan pressure curve represents a system static pressure that includes the 0.000 in. of water lab pressure. The total exhaust system pressure (fan pressure) is –0.100 in. of water (–24.9 Pa) as shown. As the lab pressure was increased from 0.000 to +0.100 in. of water (+24.9 Pa), this pressure is subtracted from the total pressure of the exhaust system and the exhaust system curve is offset by +0.100 in. of water (+24.9 Pa). This is shown in Figure 4 by the hypothetical curve originating at +0.100 in. of water (+24.9 Pa) (i.e., the lab pressure offset). The +0.100 in. of water (+24.9 Pa) offset curve intersects the fan curve to the right of the original 0.000 in. of water lab pressure system curve. Moving to the right on the fan curve shows an increase in flow rate through the system.

Thus, a positively pressurized lab would always aid capture and containment by increasing the flow rate through the hood. Conversely, a negatively pressurized lab would adversely affect capture and containment. Since the hood was tested at the threshold of C&C, any flow less than this threshold was a failure condition. The larger the negative pressure difference, the more detrimental the effect. For the lab's particular fan, a lab pressure of –0.100 in. of water (–24.9 Pa) caused failure in capture and containment because the flow rate went from 1442 cfm (680 L/s) at 0.000 in. of water lab pressure

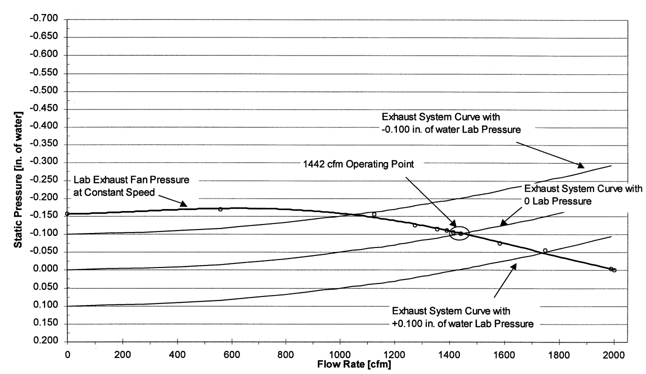

Figure 4a *Lab fan and exhaust system static pressure curves.*

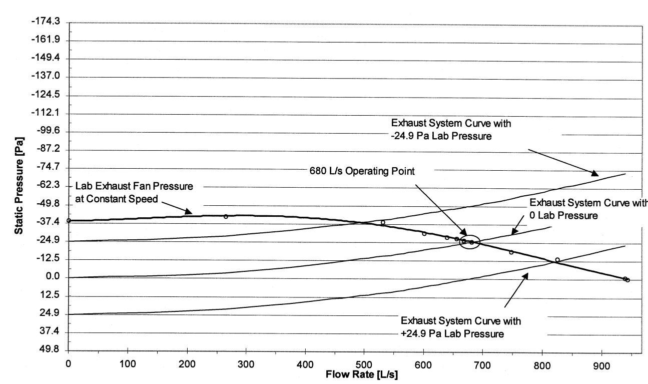

Figure 4b *Lab fan and exhaust system static pressure curves (SI Units).*

condition to 1125 cfm (531 L/s) at –0.100 in. of water (–24.9 Pa) lab pressure condition. The overall effect is that with negative pressurization, failure of the hood to capture and contain is a result of moving to a lower flow regime (left) on the fan curve due to the increase in exhaust system static pressure. At a lab pressure of –0.180 in. of water (–44.8 Pa), the flow through the exhaust system was only 560 cfm (264 L/s), clearly a spill condition.

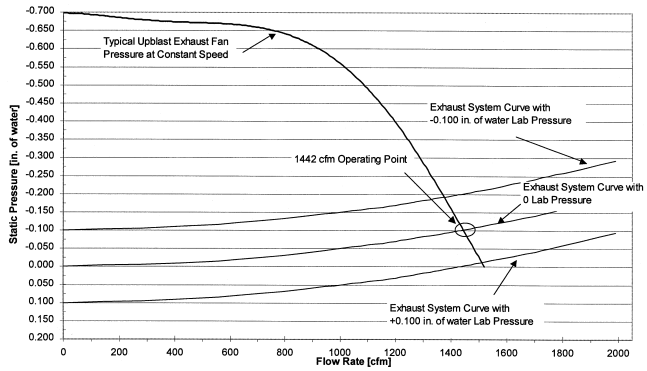

Figure 5a *Typical restaurant exhaust fan and exhaust system static pressure curves.*

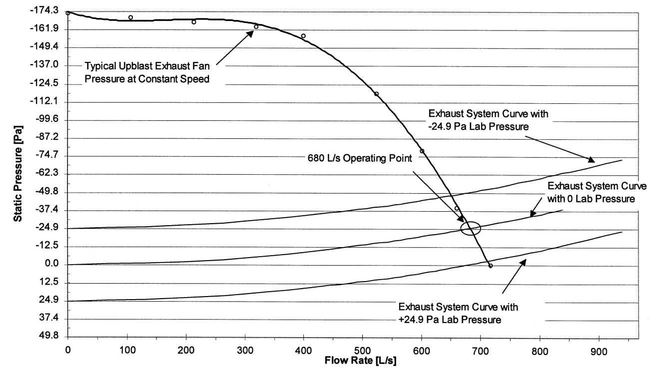

Figure 5b *Typical restaurant exhaust fan and exhaust system static pressure curves (SI Units).*

Once the interaction between the exhaust system curves and fan curve is understood, it is easy to consider more real-world scenarios. An example is shown in Figure 5.

Figure 5 shows how a typical upblast fan overlays on the previously established exhaust system curves. The proper fan speed established a pressure of 0.100 in. of water (24.9 Pa) at 1442 cfm (680 L/s). This condition included the 0.000 lab pressure. With the fan pressure curve determined, a variety of room pressure scenarios can be investigated. A neutral or correctly positively balanced restaurant with respect to ambient would ensure proper capture and containment performance of the hood. A +0.100 in. of water pressure (+24.9 Pa) increase in a kitchen would increase the flow through the hood as much as 78 cfm (37 L/s) to 1510 cfm (713 L/s). However, a slightly negative kitchen with respect to the ambient as small as –0.100 in. of water (24.9 Pa) would shift the exhaust system curve from an operating point of 1442 cfm (680 L/s) to 1375 cfm (649 L/s) and potentially create a spill condition. Note, however, that the effect is not as dramatic as demonstrated for the same negative pressure condition during the lab test due to the different characteristics of the exhaust fans. If the design exhaust rate for this upblast fan example included a small safety factor, hood C&C would be maintained. However, if make-up air dampers were closed down in trying to limit outside air or a replacement air fan failed, a more severe kitchen pressure of –0.200 in. of water (–49.8 Pa) with respect to ambient could result and yield a spill condition at 1300 cfm (613 L/s) with increased degradation of IAQ.

Focusing on the slope of a typical restaurant upblast fan curve, there are different sensitivities to pressure with respect to flow, or visa versa. If the same fan curve was examined to determine the changing slopes along the curve, three regions could be investigated. They are shown in Figure 6. If the sensitivity was calculated for these three regions as changes in flow in cfm as a result of a 0.100 in. of water (24.9 Pa) change in pressure, three values could be obtained. Region I would include system static pressures between 0.000 and +/-0.400 in. of water (+/- 99.6 Pa), or the relatively lower pressure region of the curve. For a 0.100 in. (24.9 Pa) of water change in system pressure, an 80 cfm (38 L/s) change in flow rate would

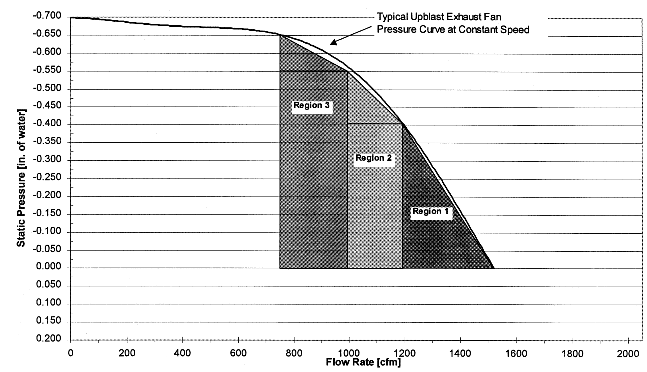

Figure 6a *Typical restaurant exhaust fan and regions of varying flow sensitivities to pressure changes.*

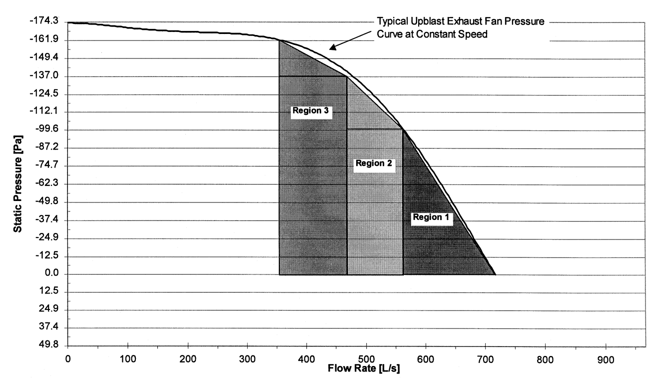

Figure 6b *Typical restaurant exhaust fan and regions of varying flow sensitivities to pressure changes (SI Units).*

occur. If we move to the more fan efficient range of Region II, the effect of a 0.100 in. of water (24.9 Pa) pressure change becomes 130 cfm (61 L/s). If this pressure change is in the negative direction and the hood was operating at the threshold of capture and containment, the change in pressure would cause the hood to fail. If we move to the high pressure range of Region III, there is a 250 cfm (118 L/s) change for every 0.100 in. of water (24.9 Pa) change in system pressure. Therefore, there is increasing sensitivity to flow, and thereby hood performance, by changes in the system static pressure as the operating point is chosen higher on the fan curve.

CONCLUSIONS

Experiment I described by this paper demonstrated that under conditions of constant exhaust fan speed, a positive pressure within the kitchen should not affect the ability of a hood to capture and contain as the increased room pressure causes an increase in the exhaust airflow. An exception to this is when the positive room pressure condition is creating a side draft or causing local disturbances within the hood area—a real-world phenomena and problem in the field. On the other hand, creating a condition of negative pressure immediately demonstrated a problem with hood performance as the airflow through the exhaust hood fell below the threshold flow rate for complete capture and containment.

Experiment II, under conditions of constant airflow and varying room pressure, confirmed (for this test setup) that room pressure did not directly impact the ability of the exhaust hood to capture and contain the thermal plume produced by the cooking process.

In real-world operation, negative pressure may or may not cause a problem with hood performance. Under conditions of constant exhaust fan speed, its effect will be to decrease airflow through the hood. However, this may not be a significant factor. For example, if the exhaust system is operating on the steep part of the fan curve, a slight increase in system pressure will only cause a slight decrease in airflow. Unless the design exhaust flow rate is at the threshold of capture and containment, spillage of cooking effluent probably will not result. However, if the system curve is intersecting the fan curve in its flatter region, the effect on airflow will be more significant. And the closer the hood is to operating at its threshold of capture and containment, the more likely the negative pressure is to cause a condition of spillage. However, if a kitchen is designed negative with respect to ambient, hood performance may not be affected. This may help to explain why the anecdotal reports on the effect of negative pressure within the kitchen are inconsistent. Simply stated: sometimes it causes a problem and sometimes it doesn't.

From a design perspective, the sensitivity of the exhaust flow rate to kitchen pressure is dependent on the exhaust fan selection and the operating point between exhaust system static pressure curve and the fan curve. Thus, designing an exhaust system to operate on the steeper part of the fan curve will minimize the impact that an undesired negative pressure condition will have on hood performance. Adding a small safety factor to the design exhaust flow rate is another strategy that will help to ensure satisfactory hood performance when a real-world kitchen turns negative. Granted, the exhaust fan speed can be modified in the field by changing sheave size and, consequently, the fan curve; however, there are limits imposed by motor current, tip speed, and brake horsepower.

In summary, the difference in pressure between the kitchen space and space outside the facility does not directly impact hood performance. Rather, it is the effect that kitchen pressure has on the exhaust system pressure drop that causes a reduction in exhaust air flow rate, possibly compromising hood performance with respect to capture and containment.

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